PATENT SPECIFICATION 1,181,526

DRAWINGS ATTACHED.

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COMPLETE SPECIFICATION.

Hydrostatic-Mechanical Power Transmission Arrangements.

We, VEB IFA GETRIEBEWERKE BRANDENBURG, of 10, Geschwister-Scholl-Strasse, Brandenburg/Havel, Germany, a corporation organised under the laws of East Germany, do hereby declare the invention, for which we pray that a patent may be granted to us, and the method by which it is to be performed, to be particularly described in and by the following statement:—

The present invention relates to hydrostatic-mechanical power transmission arrangements for vehicles, particularly agricultural tractors and working machines.

According to one aspect of the present invention, there is provided a hydrostatic-mechanical power transmission arrangement for a motor vehicle, comprising a power input shaft driving into two power branches, one of the branches comprising an infinitely variable hydrostatic transmission unit and the other branch comprising a mechanical step-wise variable gear arrangement, the branches being re-united through a power-totalling planet gear arrangement for driving a power output shaft. The hydrostatic transmission unit may be a variable transmission medium unit as defined herein.

The planet gear arrangement may comprise an internally toothed gear driven by the mechanical branch and a sun gear driven by the hydrostatic branch, a planet carrier carrying two planet gears being arranged to rotate the output shaft. Mechanical shift stages of the step-wise gear arrangement may be adapted to be shifted by means of multiple disc clutches or overrun clutches.

According to another aspect of the present invention, there is provided a hydrostatic-mechanical power transmission arrangement for a motor vehicle, comprising a first power input shaft arranged to drive a mechanical step-wise variable gear arrangement, and a second power input shaft arranged to be driven *via* an infinitely variable hydrostatic transmission unit, the power transmitted by the gear arrangement and the hydrostatic transmission unit being united through a power-totalling planet gear arrangement for driving a power output shaft. The first power input shaft may be arranged to drive the second power input shaft *via* the hydrostatic transmission unit.

To make the invention clearly understood reference will now be made to the accompanying drawing which is given by way of example and in which:—

Fig. 1 is a diagrammatic representation of the transmission, the mechanical shifting arrangement being constructed as a rotary transmission:

Fig. 2 is a similar view but with the mechanical shifting arrangement constructed as a planet gear transmission; and

Fig. 3 is a diagrammatic view of a different embodiment of the hydrostatic part of the transmission.

In the gearbox according to Fig. 1, a starting clutch 1 precedes an input shaft 2. Mounted axially parallel therewith and driven *via* gearwheels 3, 4 and a hollow shaft 5 is a hydrostatic transmission unit 6. This hydrostatic transmission unit comprises a pump G and a motor M, which are each provided with pivoting swash plates connected to the input and output shafts, respectively. According to one form of the hydrostatic transmission unit, the pivoting swash plate of the pump G bears on a backing plate which is supported on a rotatable housing of the transmission unit or on another abutment which is connected to the driven output shaft 15. At low rotary speeds, the power transmission is hydraulic. By increasing the inclination of the swash plates, the rotary speed is increased.

Gradually this results in mechanical power transmission until, at maximum obliquity, the transmission of power to the driven output shaft 15 is substantially wholly mechanical. Such a hydrostatic transmission unit will be referred to herein as a variable transmission medium unit.

Also disposed on the hollow shaft 5 and a secondary shaft 7 which is spatially displaced out of the plane of the drawing are further gearwheels 8, 9, 10, 11 of a rotary transmission. Incorporated between the gearwheel 9 and the secondary shaft 7 is a free wheel arrangement, not shown. A gearwheel 12 of a further stage is connected to the input shaft 2 by means of a clutch 13 while the gearwheel 11 is connected to the hollow shaft 5 by a clutch 14.

A driven shaft 15 from the hydrostatic transmission 6 and the driven gearwheel 11 of the rotary transmission are connected to a sun wheel 16 and an internally-toothed gearing 17 respectively. The sun wheel 16 and gear 17 are parts of a planet gear transmission system of which a carrier 19 carrying planet wheels 18 leads through a reversing gear to a differential pinion 40. The reversing gear consists of gearwheels 20, 21 with dog couplings 22, 23 and a secondary drive shaft 24 which is extended to a power take-off shaft and driven by the hollow shaft 5 through the gearwheel 25.

A free wheel arrangement 28 is braced against the motor M of the hydrostatic transmission unit 6 through a shaft 26 on a free standing transmission housing 27. This arrangement prevents the drive shaft 15 rotating backwards against the direction of drive. In this embodiment, the hydrostatic transmission unit 6 which is driven *via* the gearwheels 3 and 4 and which is a variable transmission medium unit, receives the necessary bracing moment against the free standing transmission housing 27 through the shaft 26 and drives the sun wheel 16 of the planet gearing *via* the driven shaft 15.

In the alternative embodiment shown in Fig. 3, the shaft 26 by which the motor M is braced against the transmission housing 27 is a hollow shaft, and the hydrostatic transmission unit 6 is driven through gearwheels 29 and 30. In this case, a free wheel arrangement 31 prevents a housing 32 which is rigidly connected to the driven shaft 15 from rotating backwards.

The embodiment shown in Fig. 2 has no starting clutch. The rotary transmission and the reversing gear mechanism are replaced by a single carrier planet gearing arrangement. This consists of a sun wheel 33 rigidly connected to the hollow shaft 5, double planet wheels 36 which are mounted on a carrier 34 and which roll inside a hollow wheel 35, the smaller wheels of the double planet wheels 36 meshing with a sun wheel 37 which is loosely mounted on the hollow shaft 5. The hollow wheel 35 has a brake drum which is enclosed by a brake band 38. The sun wheel 37 is provided with a further brake drum and a brake band 39. The rest of the embodiment is the same as the arrangement shown in Fig. 1. Here, too, the hydrostatic transmission unit 6 can be driven as in the embodiment shown in Fig. 3.

The transmission functions in the following way:

Starting from a standstill is purely mechanical.

In the embodiment shown in Fig. 1, the starting clutch 1 is operated, the torque transmitted to the input shaft 2, gearwheels 3, 4, hollow shaft 5, gearwheels 8, 9, secondary shaft 7 and gearwheels 10, 11 to the internal gearing 17 of the planet gear mechanism. The pivoting members of the pump G and of the motor M of the hydrostatic transmission unit 6 are so adjusted that no power is transmitted. The overrun clutch 28 acts as a non-return ratchet. Thus, the driven shaft 15 and the sun wheel 16 of the planet gear mechanism, connected to it, are stationary. Thus, the torque is transmitted directly by the internal gearing 17 through the planet wheels 18 to the carrier 19 and on to the differential pinion 40. When the starting speed has been reached, the speed can be further increased by adjusting the pivoting members of the hydrostatic transmission unit 6 to transmit drive to the shaft 15.

The rotary speeds of the sun wheel 16 and of the internal gearing 17 are, under the influence of the basic transmission ratio of the planet gear system, added together and pass to the carrier 19 through the planet wheels 18. This increase in rotary speed is possible within the entire speed range of the hydrostatic transmission unit 6, up to the maximum rotary speed of the driven shaft 15. A further increase in the number of revolutions in a given unit of time is achieved by operation of the mechanical rotary transmission by closing the shift clutch 14.

For increasing to the maximum speed at the carrier 19, the shift clutch 14 is disengaged and the clutch 13 engaged. The shift clutch 14 may be constructed as a multiple disc clutch which, once engaged, remains in engagement until manually released or it may be constructed as an overrun clutch. The overrun clutch is disengaged automatically by virtue of the higher rotary speed of the gearwheel 11 as compared with the hollow shaft 5, resulting from closure of the shift clutch 13.

In this way, all three mechanical shift stages can be operated without interrupting the transmission of power.

In the embodiment shown in Fig. 2, no clutch is required for starting. At the time of starting, the brake band 38 is tight. The

torque is then transmitted *via* the input shaft 2, the gearwheels 3, 4, the hollow shaft 5, the sun wheel 33, the double planet wheels 36, the carrier 34, the internal gear 17, the planet wheels 18, the carrier 19, to the differential pinion 40.

The driven shaft 15 coupled to the hydrostatic transmission unit 6 is stationary. It is cut in as described above only after the starting speed has been reached. Further shifting of the mechanical stages occurs after the brake band 38 has been disengaged.

In this embodiment, reverse travel is achieved by tightening the brake band 39.

In the arrangements according to Figs. 1 and 2, the hydrostatic transmission unit 6 operates with a constant delivery flow from the pump G and a variable absorption volume of the motor M while, in the embodiment according to Fig. 3, it operates with a variable delivery flow from the pump G and a variable absorption volume of the motor M. By reason of the rigid connection of the pump with the motor housing, the motor torque is fed back to the drive as an automatic thrust factor without conversion losses, and is transmitted to the driven shaft 15 by means of the pump G. Thus, the variable branch is divided into a hydrostatic part and a mechanical part, while the proportion of hydrostatic power is further reduced.

The proportion of power in the purely mechanical and in the variable branch is determined by the basic transmission ratio of the planet wheel transmission and the speeds initiated at the sun wheel 16 or internal gear 17, in proportion to the output speed at the carrier 19.

By virtue of the present embodiment, with the drive at the carrier 19, there is always a positive power flux in the planet gear transmission, so that each power branch always transmits less than the total power.

With the above described embodiments the large power needed to start the vehicle is transmitted purely mechanically which is more efficient and uses a smaller unit than when the starting power is transmitted *via* the hydraulic medium. Also the higher mechanical gear ratio may be so chosen that for the vehicle speed most used, for example, in ploughing, the power transmission is mainly mechanical.

WHAT WE CLAIM IS:—

1. A hydrostatic-mechanical power transmission arrangement for a motor vehicle, comprising a power input shaft driving into two power branches, one of the branches comprising an infinitely variable hydrostatic transmission unit and the other branch comprising a mechanical step-wise variable gear arrangement, the branches being re-united through a power-totalling planet gear arrangement for driving a power output shaft.

2. An arrangement as claimed in claim 1, wherein the hydrostatic transmission unit is a variable transmission medium unit as defined herein.

3. An arrangement as claimed in claim 1 or 2, wherein the planet gear arrangement comprises an internally toothed gear driven by the mechanical branch and a sun gear driven by the hydrostatic branch, a planet carrier carrying two planet gears being arranged to rotate the output shaft.

4. An arrangement as claimed in claim 1, 2 or 3, wherein mechanical shift stages of the mechanical step-wise gear arrangement are adapted to be shifted by means of multiple disc clutches.

5. An arrangement as claimed in claim 1, 2 or 3, wherein mechanical shift stages of the mechanical step-wise gear arrangement are adapted to be shifted by means of overrun clutches.

6. A hydrostatic-mechanical power transmission arrangement for a motor vehicle, comprising a first power input shaft arranged to drive a mechanical step-wise variable gear arrangement, and a second power input shaft arranged to be driven *via* an infinitely variable hydrostatic transmission unit, the power transmitted by the gear arrangement and the hydrostatic transmission unit being united through a power-totalling planet gear arrangement for driving a power output shaft.

7. An arrangement as claimed in claim 6, wherein the first power input shaft is arranged to drive the second power input shaft *via* the hydrostatic transmission unit.

8. A hydrostatic-mechanical power transmission arrangement for a motor vehicle constructed, arranged and adapted to operate substantially as hereinbefore described with reference to and as illustrated in Fig. 1 of the accompanying drawings.

9. A hydrostatic-mechanical power transmission arrangement for a motor vehicle constructed, arranged and adapted to operate substantially as hereinbefore described with reference to and as illustrated in Fig. 2 of the accompanying drawings.

10. An arrangement as claimed in claim 8 or 9, modified substantially as hereinbefore described with reference to and as illustrated in Fig. 3 of the accompanying drawings.

11. An agricultural vehicle comprising a hydrostatic-mechanical power transmission arrangement as claimed in any one of claims 1 to 10.

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1 SHEET *This drawing is a reproduction of the Original on a reduced scale*

Fig.1

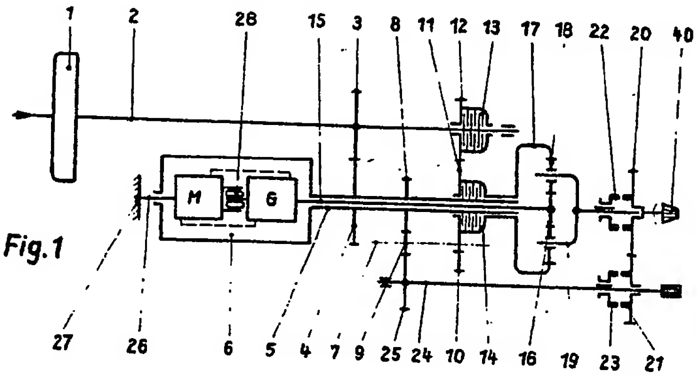

Fig.2

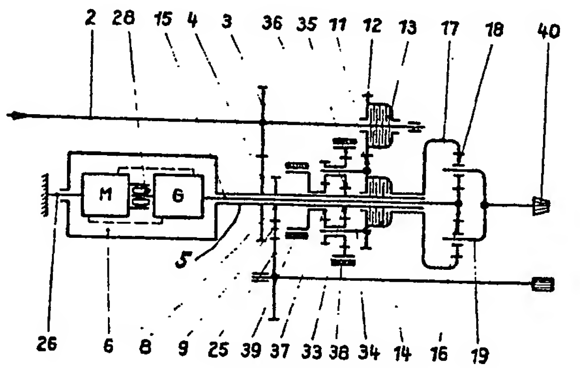

Fig.3